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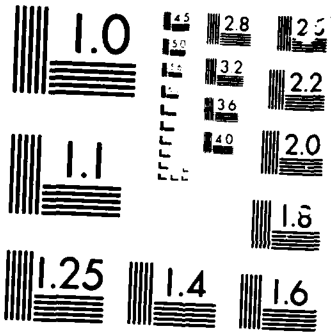


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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The research has resulted in additional calculations which will provide essential guides to future work. Using the Shanley model as a cheap and effective tool, calculated results show the influence of the geometrical parameter eta (= thickness of sandwich beam/half span) and of damping. It is seen that the anolmalous behavior occurs only in a small range of thickness.

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20. ABSTRACT CONTINUED

Damping changes the response behavior fundamentally: counter-intuitive behavior still occurs but in a much more irregular manner, suggestive of chaotic behavior. Calculations using ABAQUS for the pinned uniform beam of "standard" dimensions and for a "test specimen" of slightly different dimensions show the characteristic diagrams for the two cases. It is surprising that qualitatively different shapes are obtained for the critical range of loads. The anomalous behavior being studied is evidently sensitive to parameters in unexpected ways.

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Subject:

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"Feasibility Study of Test Problems for Finite Element/Finite Difference Programs for Structural Dynamics"

"Feasibility Study of Test Problems for Finite Element/Finite Difference Programs for Structural Dynamics"

Principal Investigators: P.S. Symonds and H. Kolsky (Brown University)

## 1. Background

The phenomenon here studied consists of anomalous computed response of certain structures to pulse loading (as in blast or impact). The example discussed in the initial publication describing it [1] (attached herewith) is a beam whose ends are attached to smooth fixed pins and which is subjected to a rectangular pressure pulse. When the material behavior is assumed to be elastic-perfectly plastic, and the response is calculated by a finite element or finite difference program, it is found that different programs may predict grossly different final deformed states, although they generally agree closely with respect to the first peak deflection, reached shortly after the termination of the pulse. Following the peak deflection there is a recovery, which is initially elastic but which may involve further plastic deformation. Eventually plastic flow is completed, and the structure then undergoes an elastic vibration which continues indefinitely in the absence of damping. For certain values of the parameters (of structure geometry, material properties, and load pulse) the computed final permanent deflection may depend with extreme sensitivity on these parameters, and the predicted final displacement may be either in the direction of the loading or in the opposite direction. The prediction is sensitive also to details of the computation (e.g. type and size of finite elements, type and step size of step-by-step time integration, etc.).

The sensitivities observed can be traced to dynamic instabilities associated with combinations of axial compressive and flexural stress states which are developed. Their potential practical importance lies in the fact that they may occur in industrial design calculations and may be the source of large errors or uncertainties.

An efficient tool for clarifying the essential features of these complex phenomena is a one-degree of freedom (Shanley) model consisting of rigid bars connected by a deformable cell whose properties are taken as those of a sandwich beam. With appropriate numerical values, this can reasonably represent the uniform beam. This model was introduced in [1], and it was found that the counter-intuitive response calculated for the uniform beam for a particular load pulse was observed in the model for a small range of loads. Further calculations [2] (attached) showed that for the model there were three critical load values at which discontinuous transitions in response behavior occurred, and significant properties of these critical values were illustrated by phase plane plots and diagrams of the period of the continuing elastic vibration. It was shown in [2] that the unavoidable errors associated with finite time step in standard numerical methods can have the same effect as a change of loading, in the critical region.

## 2. Recent Results

In recent months, aided by ARO Contract Prop. No. 23225-EG we have (A) made additional calculations for both a Shanley model and a uniform beam (using the multi-purpose program ABAQUS); and (B) made laboratory specimens and performed preliminary impact tests. The results will be summarized in turn.

(A) <u>**Calculations**</u>

In the calculations, the guiding concept has been that of the characteristic diagram which shows the maximum and minimum displacements of the continuing elastic vibration as functions of the pulse load magnitude. When the Shanley model is used, the initial peak deflection can be taken as an equivalent "load" magnitude, for simplicity. In our Shanley model, there are two dimensionless parameters $\mu = E'/\sigma_o$, $\eta = h'/\ell'$, where $E'$, $\sigma_o$, are respectively, the Young's modulus and the yield stress of the flanges of the sandwich beam, and $h'$, $\ell'$ are the distance between the flanges and the half-span of the model, respectively. To make the model correspond adequately to the prototype, we chose $\mu$=400, $\eta$=0.0271, for reasons outlined in [1]. Figure 1(d) shows the characteristic diagram for this original model. Here the "slot" represents the counter-intuitive behavior. The diagram shows three critical values where the behavior undergoes discontinuous change. The other five diagrams in Fig. 1 show how the response behavior changes as the geometrical parameter $\eta$ is changed. There is no slot if $\eta$=0, i.e. when the model cannot exert a bending moment. The slot is widest at $\eta \cong 0.02$, but is not present for $\eta$=0.03 and 0.04. These results show that the counter-intuitive behavior is associated with plastic deformation in bending, but does not occur if the bending action is too strong relative to that of axial force.

These results are relevant to questions as to the dimensions of beams or plates such that similar dynamic instabilities occur. Other parameters of the Shanley model also need to be studied as a guide to investigation of practical structures. We have so far looked only at one other physical parameter of potential importance. This is the role of damping. Various kinds of damping and friction effects must be present in any real structure. We have considered linear viscous damping as the simplest starting point, but even the preliminary results show that it introduces a considerable additional complication.

Defining the strength of damping by the ratio $\zeta=C/C_{cr}$, where C is the damping coefficient and $C_{cr}$ is its critical value for small amplitude vibrations, we have used several values, but confine attention here to $\zeta$=0.05 (i.e. "5 percent of critical"). It is immediately seen that the behavior of interest here is qualitatively different than when $\zeta$=0. Figure 2(a) shows one example. Here the initial deflection is such that if damping is zero, the continuing vibration involves snapping through between positive and negative extreme values. With damping, the plus-minus vibration changes gradually to a plus-plus vibration. There is no sharp distinction between the transient motion and the continuing elastic vibration. Another different behavior is shown in Fig. 2(b), for larger values of initial displacement. Here the small change in $\phi_o$ from 0.0959 to 0.0960 produces a change from a minus-minus to a positive-positive vibration. (No transition of this type occurs for $\zeta$=0). A new definition of $\phi_{max}$ and $\phi_{min}$ is needed, in view of the behavior illustrated in Fig. 2(a), in order to plot a characteristic diagram analogous to those of Fig. 1. For example, a particular instant $t_*$ may be chosen, and $\phi_{max}$ and $\phi_{min}$ taken as the maximum and minimum deflections following that instant. Taking $t_*$ = 2 msec and 5 msec, the resulting diagrams are shown in Figs. 2(c) and 2(d), respectively.

These examples serve here to show that the phenomena we are considering, associated with dynamic instabilities, become substantially more complex when damping is included.

Finally, our calculations have included ones for the prototype pulse-loaded uniform beam, again with the main objective of determining the characteristic diagram. The load

temperature. They were then precipitation hardened for 9 hours at 155°C. With this treatment their stress-strain behavior was approximately perfectly plastic, with yield stress about 40 ksi.

The beams fixed in the frame were mounted in a "Hyge" shock testing machine which enabled them to be impacted by a fast moving hammer. The "Hyge" machine consists essentially of an air gun which activates a "hammer" which then runs along parallel rails at velocities between 10 ft/sec and 150 ft/sec. In our set-up we have used two hammers; when actuated, the air gun propelled the first hammer which hit the second hammer, placed close to the beam specimen, and this impacted the specimen. Two hammers were used so as to enable measurements to be made of the force-time history of the impact. This was done by fixing strain gauges to measure axial strains on the cylindrical nose of the second hammer, and since this second hammer travelled only a short distance, electrical leads could easily be run from it to the recording oscilloscope.

Other methods were used at the same time for monitoring the impact. We recorded the impact by a "Fastax" high speed "cine" camera. This camera enables pictures to be taken at speeds up to 8000 frames/second., but since the time period it was necessary to cover was comparatively long, the camera was used at considerably lower framing speeds. The velocity of the second hammer immediately prior to impact was measured by arranging for this hammer to make two electrical contacts a known distance apart just before impact, and measuring the interval between the signals with a microsecond timer. A still photograph of the beam was taken immediately after impact, and the deformed beam was measured after it had been removed from the machine.

Figure 4(c) shows photographs of typical tested beams with small; moderate, and large permanent deformation. The third case illustrates the main difficulty of the set-up, namely that of fixing the axle rods so that they do not approach each other during the impact. This was evidently not achieved in the first tests, as shown in the photograph. In subsequent tests a better means of wedging the axle rods in place was used, and their relative displacement was much reduced. As a simple measure of load pulse magnitude, we used the permanent displacement after the impact. (A better measure is the peak displacement reached during the impact; this would be used in future more complete experiments). The present tests included a series of 6 tests in which the axial constraint was considered reasonably satisfactory. In this series the impacting velocity was gradually reduced to a value such that the permanent displacement was barely measurable. If other conditions were satisfied, the counter-intuitive behavior (negative permanent displacement) should have been observed at one of the intermediate impact velocities, as indicated by Fig. 3(b). No final deformations in the negative direction were obtained, however.

In the relatively crude tests we were able to make in this program, it was not possible to check that the other necessary conditions were satisfied. The most critical conditions are perhaps the "fixed pin" end constraint, and loading by a pulse short enough to have negligible effect during the recovery phase. It is possible that a very small deviation from complete fixity would be sufficient to eliminate the counter-intuitive behavior. The nagnitude and type of friction at the supports may be critical. These conditions will be a focus of future work. The load pulse also may not satisfy the necessary condition, perhaps being too long because of multiple

contacts. This would prevent the recovery displacement from swinging into the negative range. The duration of contact will be shortened in future work.

## 3. Summary and Conclusions

The contract has allowed making additional calculations which will provide essential guides to future work. Using the Shanley model as a cheap and effective tool, calculated results are shown in Figs. 1 and 2. These show the influence of the geometrical parameter $\eta$ (= thickness of sandwich beam/half span) and of damping. It is seen that the anolmalous behavior occurs only in a small range of thicknesses. Damping changes the response behavior fundamentally: counter-intuitive behavior still occurs but in a much more irregular manner, suggestive of chaotic behavior. Calculations using ABAQUS for the pinned uniform beam of "standard" dimensions and for a "test specimen" of slightly different dimensions are shown in Fig. 3. These show the characteristic diagrams for the two cases. It is surprising that qualitatively different shapes are obtained for the critical range of loads. The anomalous behavior being studied is evidently sensitive to parameters in unexpected ways.

The experimental work done was enough to provide experience in fabricating a pin-ended specimen with a frame to provide fixed-pin end conditions. The Hyge air gun apparatus was shown capable of providing impacts over an adequate range of speeds. Full instrumentation was not attempted; high speed moving pictures taken by a Fastax camera provided approximately 20 photographs during and subsequent to the impact. No tests in a series considered most reliable showed anomalous final deformations. Several modifications are suggested as desirable to more nearly reproduce the conditions presumed in the calculations. With these and more complete instrumentation, the presently designed specimen and apparatus appears to provide a feasible scheme for investigating experimentall the instabilities and anomalous behaviors of present interest. Further calculations must guide the anticipated experiments to further investigate sensitivities to the parameters of the structure and test set-up. These must include both calculations on actual structures using a finite element program (ABAQUS), and ones for the much simpler Shanley-type model.

The question of defining useful test problems, taking advantage of the observed sensitivity of computed results to details of the code and solution technique, has not yet been resolved. The characteristic diagrams of Fig. 3 are relevant to this. They show why qualitatively different solutions are to be expected from different codes and techniques when the load is near a transition region. For example, the case $P_o$ = 19.2 kN/m, as used in the problem treated in [1, 2], is near such a region. The determination of the "correct" solution for a load $P_o$ in the range (19.2, 19.8) kN/m has not been investigated, and in fact the existence of a unique "correct solution" must be questioned. This remains to be investigated. The relation of test results to computed solutions, which presumably will require more realistic and complex characterization of structural and loading parameters, also remains to be investigated. Without considerable further research, the class of problems here being studied allows interesting comparisons to be made between different computer codes and strategies, but does not yet provide test problems in the sense of accepted standards of comparison.

## 4. Personnel

The work here described involved, in addition to the principal investigators, participation of J.F. McNamara (on leave from University College, Galway), Francesco Genna (on leave from Politecnico di Milano, Milan) and C.W. Frye (graduate student at Brown University).

## 5. Publications

A paper on the influence of slenderness ratio and friction on dynamic plastic instabilities in response to pulse loading is in preparation by P.S. Symonds and Francesco Genna. One on characteristic diagrams for pinned beams will be prepared by P.S. Symonds and J.F. McNamara, after further calculations are completed.

## References


1. Symonds, P.S. and Yu, T.X., "Counter-Intuitive Behavior in a Problem of Elastic-Plastic Beam Dynamics," J. Applied Mechanics, Vol. 52, pp. 517-522, September, 1985.

2. Symonds, P.S., McNamara, J.F. and Genna, F., "Vibrations of a Pin-Ended Beam Deformed Plastically by Short Pulse Excitation," from *Material Nonlinearity in Vibration Problems* ASME publication AMD-Vol. 71, Editor, M. Sathyamoorthy pp. 69-78, 1985.

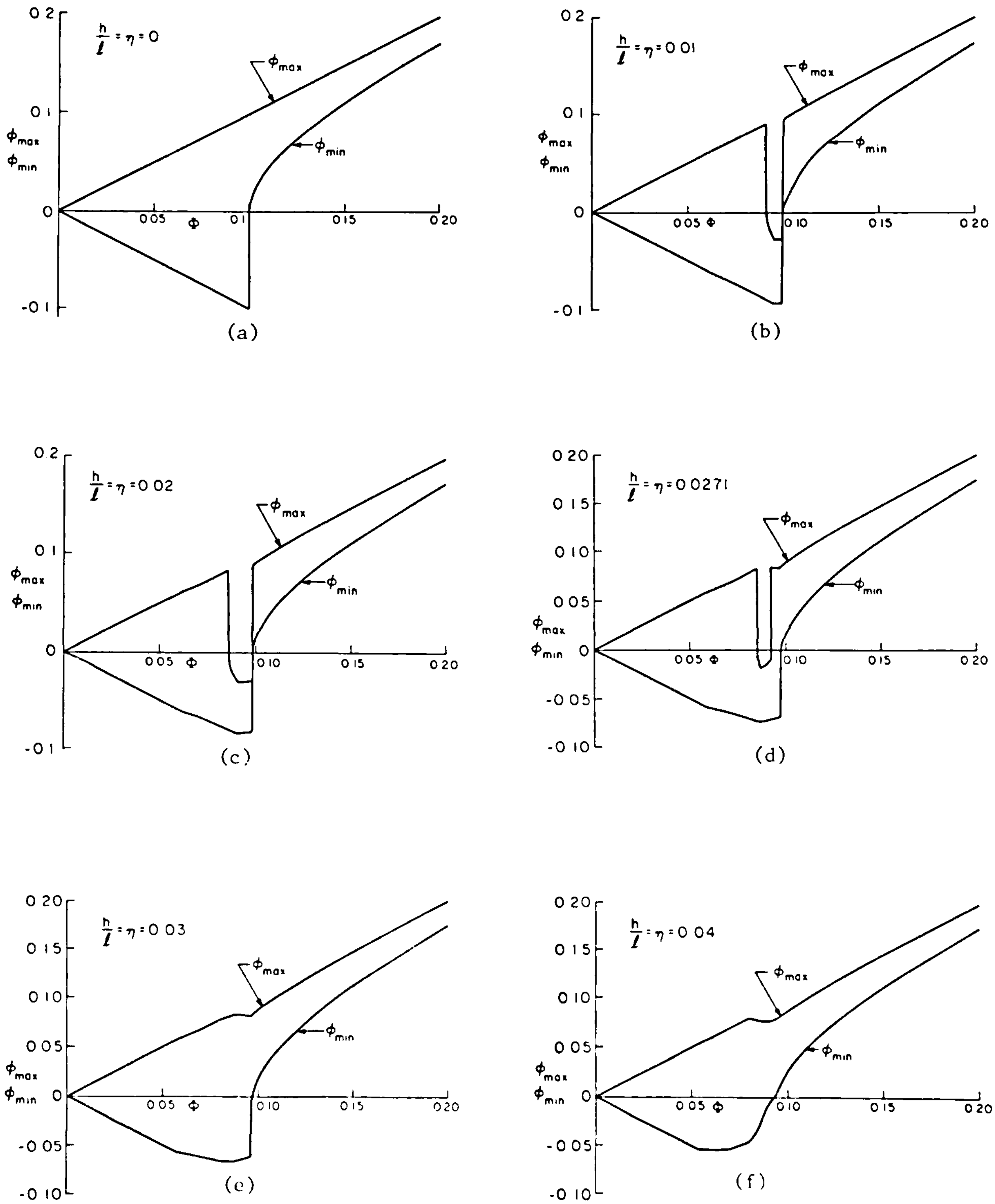



Figure 1. "Characteristic diagrams" for the Shanley model showing effect of changing parameter η=(flange separation) (half span). Diagrams show bounds on continuing elastic vibration as function of starting (peak) deflection.

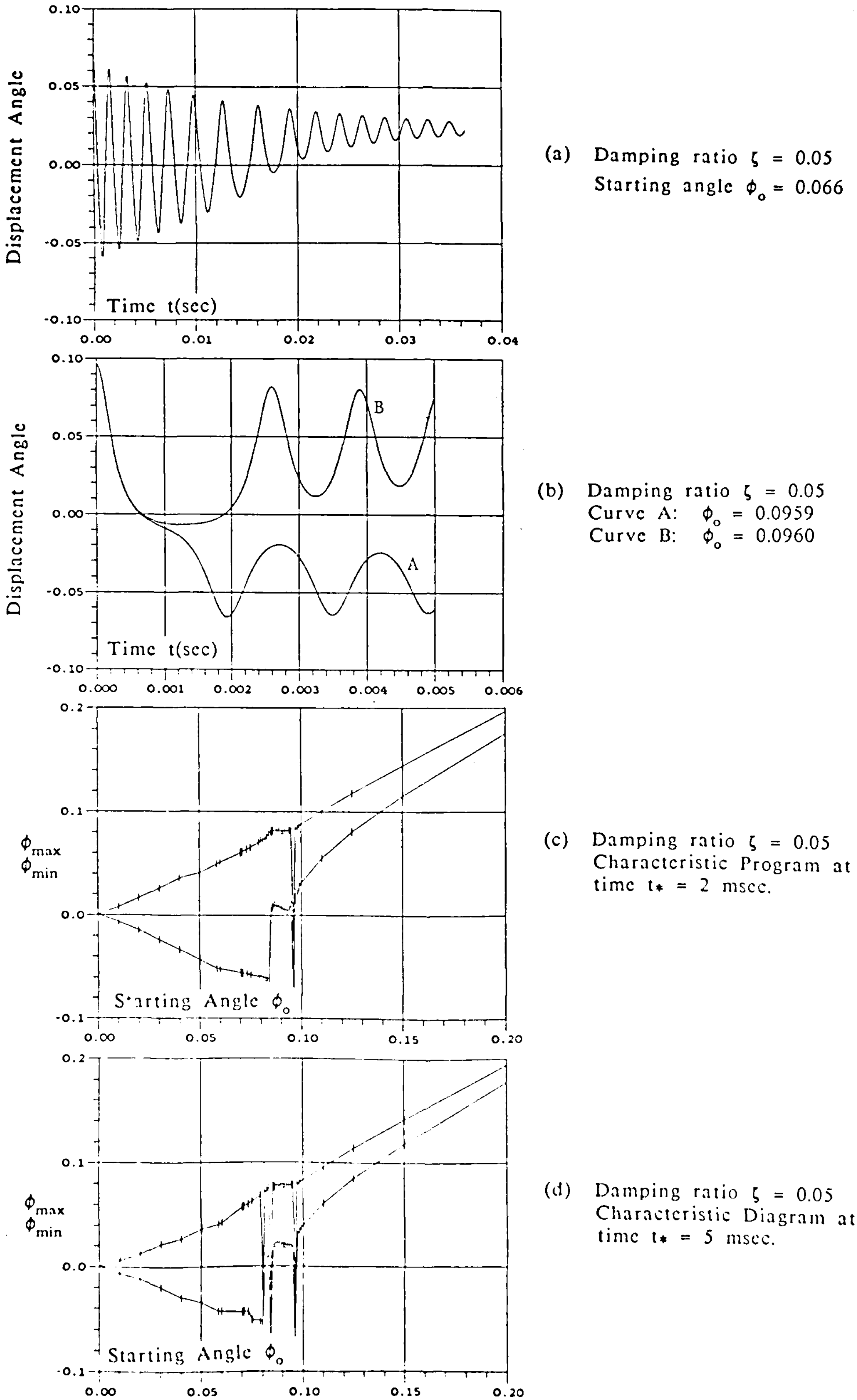
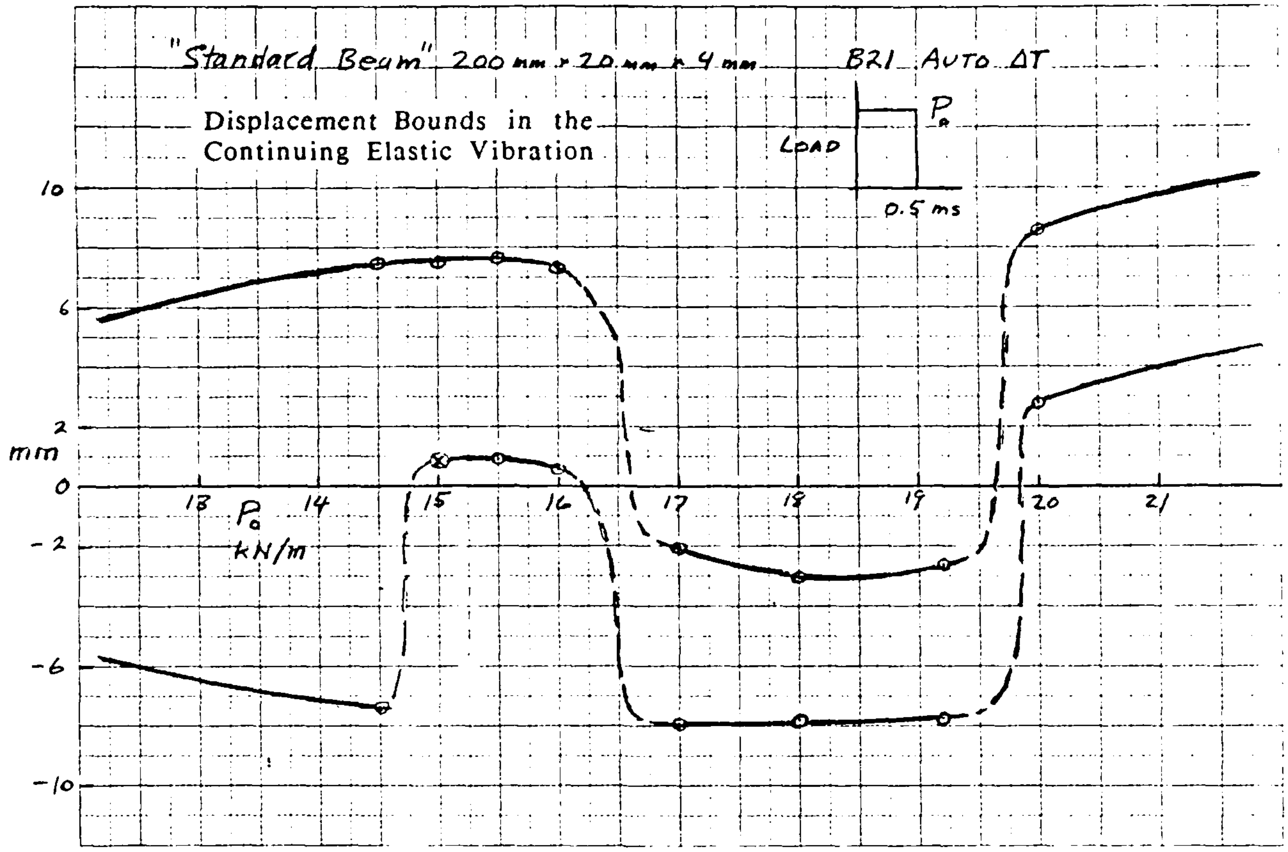
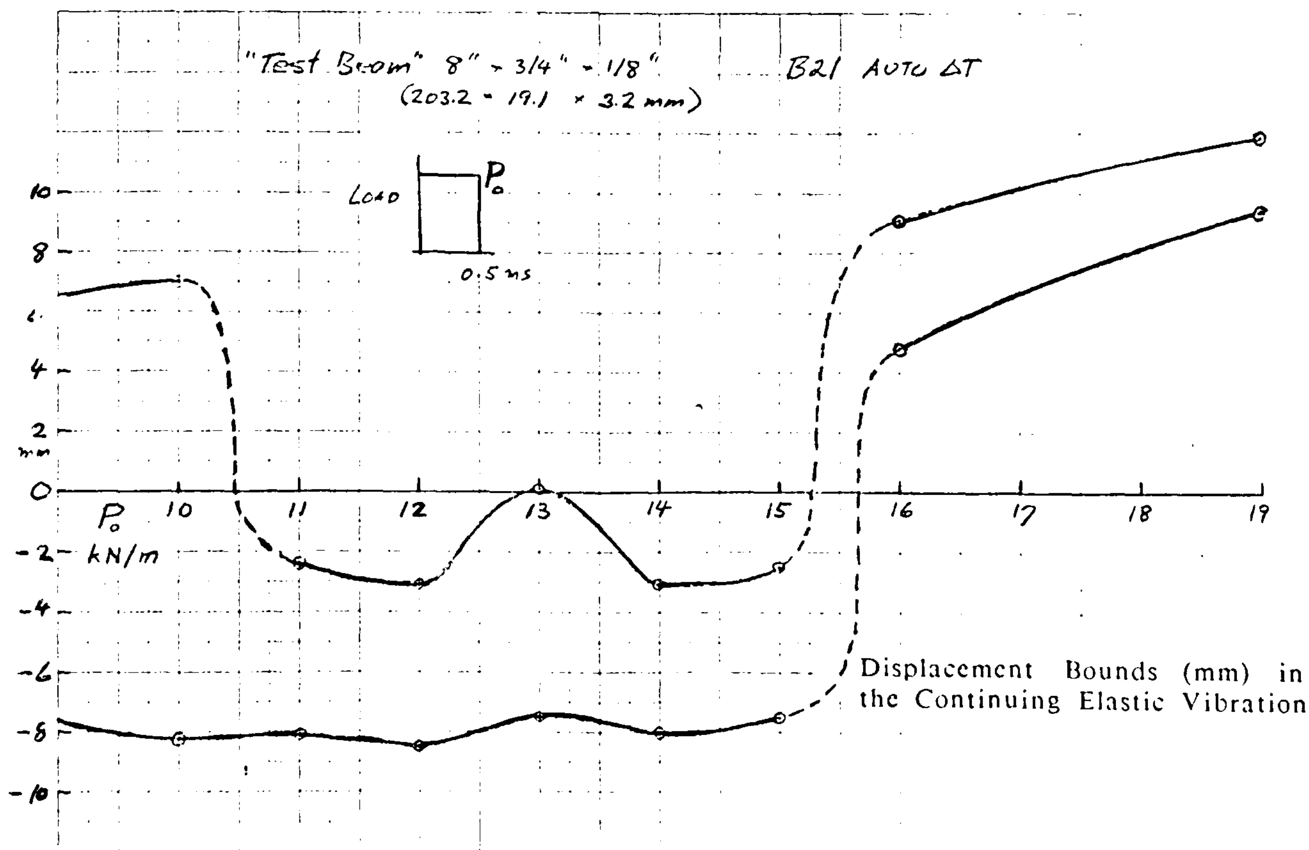



Figure 2. - Diagrams showing effects of damping for $\zeta = C/C_c = 0.05$, $\mu = 400$, $\eta = 0.0271$.

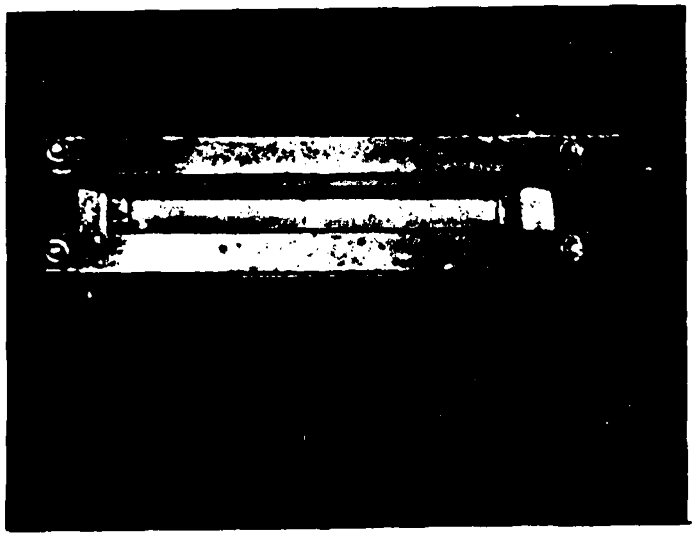



(a)




(b)

Figure 3. - "Characteristic diagrams" (by ABAQUS calculations) for two pin-ended uniform beams: (a) with dimensions as used in [1,2]; (b) with dimensions used in laboratory test specimens.



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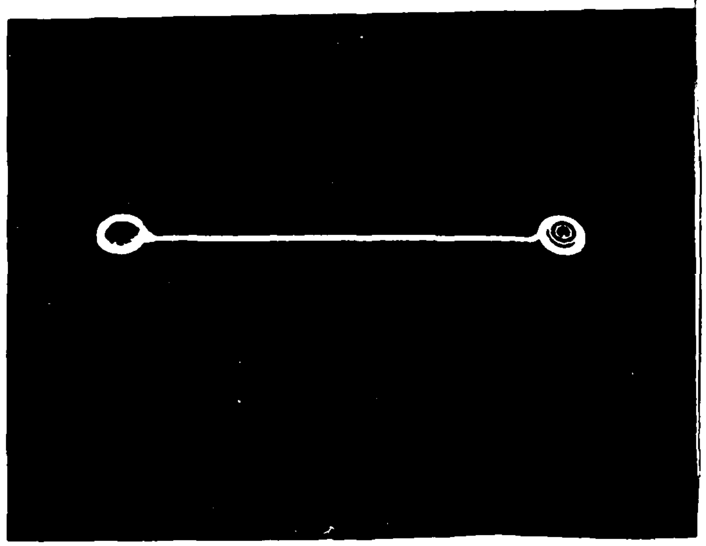

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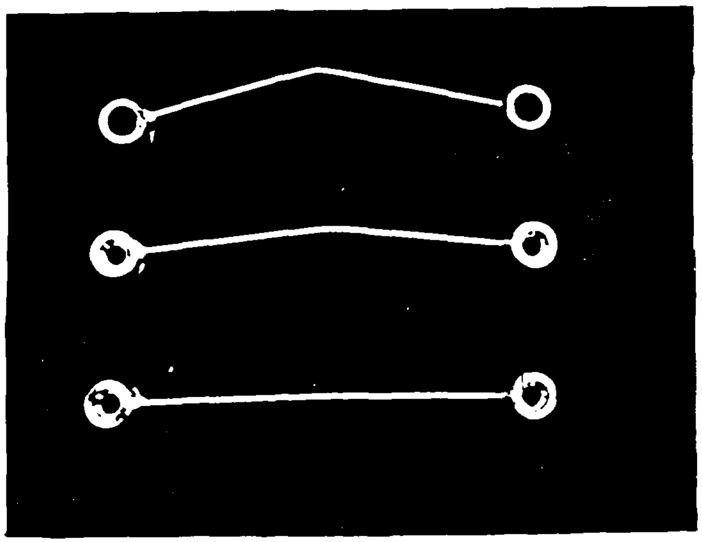


(c)

Fig. 4 (a) Beam in Frame; (b) Typical Beam Specimen; (c) Typical Specimens after impact test.

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